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A NEW DYNAMOMETER BRAKE

By

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The measurement of power furnished or utilized by engines and the resulting torques, speeds, etc., has, in recent years, acquired an ever-increasing importance, because it has become a valuable means of scientific and technical research, not only in investigations of a more or less economic character (efficiency, etc.), but also in connection with the mechanical functioning of the engines themselves. These measurements constitute a valuable guide for inventors and constructors, especially in all cases where quantitative calculations cannot be made with sufficient accuracy for attaining the desired results.

The mechanism here described belongs to the class of dynamometer brakes in which the motive power is transformed into heat in the brake itself.

This mechanism was invented by the writer for the purpose of measuring forces in which the two factors, torque and speed, vary within broad limits, the mechanism itself being of simple construction and of still simpler operation.

The results listed below were obtained with an instrument made by the Experimental Institute of Aeronautics for its own laboratories and show to what extent I accomplished my desire to make a modest contribution to the solution of this important prob-

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lem.

Description and Operation.

The dynamometer brake under consideration consists essentially of (Fig. 1):

Three wheels, A, B_1, B_2, geared together, the central wheel A being coaxial with and solidly attached to the shaft of the engine being tested;

A case C (oscillating between two stops pp) containing said wheels, which are free to rotate with slight friction at the ends of the teeth and on their faces;

Two pipes t_1 and t_2 each with one outlet (e and f);

A cock R, called the regulator, which serves to obstruct more or less the outlet of the pipe t_2;

An arm V forming a continuation of the case C and carrying a variable weight P;

Lastly, a tank S containing some liquid (water and glycerine, soapy water, oil, etc.) with or without some means for cooling the liquid.

If the wheel connected with the engine shaft is now made to rotate, while the end e of the pipe t_1 dips into the tank S, the system will evidently function like a double geared pump, so that (assuming the rotation directions indicated in the figure) a suction will be generated in the small chambers a a and the liquid will flow through the pipes t_1, while said liquid will be compressed in the chambers b b and will therefore tend to pass through the regulator R and the outlet f.

If the regulator is wide open, the resisting torque of A is quite small and due exclusively to mechanical friction and that of the liquid in the pipes t_1 and t_2. If the regulator is completely closed, the resisting torque is very large (theoretically infinite) and increases with the tightness of the gears in the case. If the opening of the regulator is kept between said limits, the torque also remains between the corresponding limits (theoretically 0 and ∞).

Let us designate by F_1, F_2, F_3 .... and C_1, C_2, C_3, ... the absolute values of all forces and torques which can originate in the chain of wheels, including the bearings and the case, and let us consider between these forces and torques (which all have a double sign), exactly what ones act on the case.

If we reduce all the forces to O, they will be balanced by the reaction of the supports on the case and the torques generated by changing C' C'' C''', etc., added to C_1 C_2 C_3 etc., will be, in the condition of dynamic equilibrium (n = constant) which alone interests us, such that

$$\Sigma C + \Sigma C' = \text{motive torque},$$

that is, the case is always subjected to a resultant torque identical with the motive torque.

As for the forces acting on the case, it is evident that, for the symmetry of the system, we must have $\Sigma F = 0$.

It follows that the only cause capable of generating a torque, not measured by the weight P, is the weight of the brake, which produces a torque of friction on the pivots of the case. These

pivots rest on ball bearings which form part of the foundation. This torque (which must be considered at the moment the motion begins and precisely when the case tends to assume a horizontal position) is very small and wholly negligible. The conditions are the same therefore in this case as for dynamic, hydraulic and other brakes. It is evident that here also the torque must be measured when the arm V is exactly horizontal.

Calling

P the weight applied to the end of V,

l the distance of the center of gravity of P from the vertical line passing through O,

n the r.p.m. of wheel A,

N the motive power in HP, we have:

(a) $$N = \frac{2\pi n P l}{60 \times 75} = k P n, \text{ in which } k = \frac{2\pi l}{60 \times 75} = \text{constant},$$

since the position of the weight P remains unchanged, It might be more convenient, in some cases, to keep the weight P constant and vary the length l of the arm V, in which event formula (a) would be replaced by

(b) $$N = k' l n, \text{ in which } k' = \frac{2\pi P}{60 \times 75} = \text{constant}.$$

Evidently there are always two quantities (weight and speed) to be measured, the same as with other dynamometer brakes.

Construction Details.

Figs. 2 and 3 were taken from the original drawings of the brake experimented upon. The wheels R are made of nickel steel,

machined but not polished. They have each 40 teeth and their pitch diameter is about 80 mm. Their faces have concentric circular grooves (L) for reducing the pressure of the liquid in passing from the circumference toward the center. This is advantageous, not only with reference to the strength of the case, but also to the tightness of the stuffing boxes p of the center wheel. All gear shafts, as well as those of the case, are provided with ball bearings (S,s).

The plane passing through the axes of the wheels is brought into the horizontal position, before starting, by moving the counterpoise C until the checking points c c are opposite each other. The weight P is suspended by a rod A terminating in a circular knife edge which rests in a circular groove in the shank of the screw V which serves for regulating the length of the brake arm.

A damping piston D is fitted into the base B and functions with either air or oil, according to the desired damping effect.

K is the regulator screw with 24 threads per inch. During the experiments, the head of this screw was provided with a scale, rendering it possible to determine, by referring to a stationary index, the number of turns made by the screw counting from the completely closed position. This number of turns was accordingly called the "degree of opening of the regulator."

The brake is attached to the engine by means of the flange G.

The constant K of formula (a), with $l = 285.7$ mm., now becomes

$$K = \frac{2 \times 3.14 \times 0.2857}{60 \times 75} = 0.0004,$$

and the power is given by

$$N = 0.0004\ P\ n.$$

In order to convey some idea of the sensitiveness of the brake to variations of the weight P, we will add that, when in working order, 2 grams placed on the end of the arm suffice to move the checking point and that 3 grams will cause a displacement of about 2 mm.

Theoretical Torque and Power.

We will now work out a few formulas regarding the operation of the brake, on the assumption that the gear wheels within the case are perfectly tight and that all frictional resistances are negligible. Then, by comparing the experimental with the calculated results, we shall see how the actual functioning differs from the theoretical. Numerical values always refer to the I.S.A. brake.

The resisting torque evidently does not have a constant value, but one that varies periodically between a maximum and minimum, the oscillation period depending on the pitch and speed of the gear wheels.

Let us call

p = pressure of liquid in kg/cm.^{2}

l = width of teeth = 20.00 mm.

b = height of teeth = 4.32

r_e = outside radius of gear wheel = 42.00

r_p = pitch " " " " = 40.00

r_i = root " " " " = 37.68

Let us consider the case where the contact point b (Fig. 1) of the engaged teeth lies on the straight line connecting the wheel centers oo'. The resisting torque of the wheel O, due to the wheel O', is given by

$$(1) \qquad C_r = p\,l\,(r_e - r_i)\,\frac{(r_e + r_i)}{2} - (r_p - r_i)\,\frac{(r_p + r_i)}{2} + Pr_p;$$

from which, by considering the equilibrium of the wheel O', we obtain

$$(2) \qquad Pr_p = p\,l\,(r_e - r_i)\,\frac{(r_e + r_i)}{2} - (r_p - r_i)\,\frac{(r_p + r_i)}{2};$$

or, by substituting expression (2) in (1)

$$(3) \qquad C_r = 2p\,l\,(r_e - r_i)\,\frac{(r_e + r_i)}{2} - (r_p - r_i)\,\frac{(r_p + r_i)}{2} =$$

$$= 2p\,l\left[\frac{r_e^2 - r_i^2}{2} - \frac{r_p^2 - r_i^2}{2}\right] = 2p\,l\left[\frac{r_e^2 - r_p^2}{2}\right]$$

By substituting the numerical values in (3) we obtain

$$C_r = 2 \times p \times 20\left[\frac{42 \times 42 - 40 \times 40}{2}\right] = p \times 3280 \times 10^{-5} \text{ kgm.} =$$

$$= 0.03280\ p \text{ kgm.}$$

Calling C_c the total resisting torque due to the two driven wheels, we have $C_c = 2\,C_r$ and we will call C_c the <u>mean con-</u>

ventional torque, which, though not coinciding with the mean true torque, approximates it very closely. It is not convenient to consider the latter here, since it is not possible to obtain a simple expression for it, by varying it with relation to many other quantities of a constructive character (aside from l, r_e and r_p, which alone figure in equation 3), while, on the other hand, we are only interested in being able to make comparisons with the experimental values of the torque. In the cases we have investigated, the difference between C_c and the mean true torque is about 1.5% of the latter, evidently quite a small quantity.

The power corresponding to C_c is:

(4) $$N = \frac{2\,\pi\, n\, C_c}{60 \times 75} = 0.0000915\ np;$$ in which n = the r.p.m. computed from the middle wheel. We will call N the mean conventional power.

It should here be noted that, considering the periodic character of the torque, it is advisable to use gear wheels with unequal numbers of teeth, so that the resisting torques of the two driven wheels will be in quadrature with reference to each other, by which means the resultant torque is rendered considerably more uniform. Moreover, the driven wheels do not need to be of the same size as the middle wheel and only two in number, but may be smaller than the latter and more in number, so as to obtain larger and more uniform torques. Naturally, in this event, the pipes issuing from the little chamber, where the liquid is drawn in and compressed, still unite in two pipes arranged as in Fig. 1, with

only one regulator.

Experimental Results.

A series of experiments was carried out in order to ascertain the characteristics of the brake. The brake was attached to a small electric engine with compound excitation by means of a spherical pin joint. The current was obtained from a storage battery and, by using a larger or smaller number of cells, it was possible to regulate the voltage. The speed and torque were obtained for each tension by operating two rheostats: one inserted in the feed circuit and the other in the shunt excitation circuit.

The liquid employed was oil, which was cooled by immersing in it a tubular radiator through which flowed water at about 15.5°C (60°F), the rate of flow being capable of variation, in order to regulate the temperature of the oil. The distance of the axis of the middle wheel from the surface of the oil was about 250 mm. The oil used was "Ambroil F" chosen for its purity, its low viscosity and especially because it did not form persistent air bubbles, as was the case with the other oils tried. Its characteristics are:

Engler viscosity 16.1 at 20°C.
3.6 " 50°C.
Flashing point between 200 and 205°C.

The speed was measured with a Tel tachometer; the pressures, with a Bourdon manometer accurately adjusted in advance.

The functioning was generally very satisfactory, both from the point of view of uniformity and of sensitiveness to adjust-

ments of the regulator. At the lowest speeds (below 100 r.p.m.), there was a slight oscillation of the checking point about its horizontal position, but this did not interfere with the accuracy of the readings. Without doubt, the previously mentioned expedient of employing gear wheels with an unequal number of teeth and a slight increase in the size of the damping piston would have eliminated this phenomenon, which was eliminated automatically at a higher speed by the inertia of the oscillating group, consisting of the case, gear wheels, weight, etc.

Table I gives the results of the above experiments, which served for tracing the curves of Fig. 7.

During each experiment, the regulator opening was kept the same and also, within the limits permitted by the oil-cooling device the temperature of the inflowing oil, so that, by varying the speed, the other quantities measured (torque, power, pressure and temperature of outflowing oil) varied only with reference to the speed, hence giving the characteristic functions of the brake.

Let us now examine the two most important of these functions, torque and power, through the group of curves of Fig. 7.

T o r q u e.

We immediately note the different behavior of Curve I, corresponding to the second degree of the regulator opening, and of Curves II, III and IV, corresponding to the degrees 5, 10 and 15, the first being concave and the others convex. These curves approach the corresponding values of the pressures which, as shown

on Table I, clearly increase in experiment I and decrease in the others. On the other hand, the reason for the greater torque in experiments II, III and IV, in which the pressure decreases, exists mainly in the friction between the oil, gear wheels and case. In any event, there is the self-evident opportunity to reduce the regulator opening, when it is desired to increase the torque. This holds good for all speeds.

P o w e r.

The shape of the power curves is similar in the four cases considered. It is always a question of increasing curves in which the mean angular coefficient of the tangents, within equal speed limits, continues to increase as the opening in the regulator is diminished, or, with increasing speed, the rate of the power increase is augmented by diminishing the regulator opening. This effect is especially noticeable for the smaller regulator openings. It is rendered most evident by the comparison of Curves I, II and III, corresponding to the openings 2, 5 and 10.

P r e s s u r e.

As already observed, the behavior of the pressures also varies according to the size of the regulator opening, so that, while with opening 2, for example, the pressure increases a little less than the speed, with openings 5 and 10 it varies inversely as the speed. This does not therefore verify the theoretical law of pressure variation with relation to the angular speed ω, which should be

$$p = K\,\omega^2 \text{ (in which K is constant)},$$

which is easily verified by applying the theorem of Bernoulli between the small chambers, which are under the pressure p, and the outlet of the regulator. It is interesting anyway to note that, even with decreasing pressure the torque and power always increase (experiments II and III).

Temperature of Outflowing Oil.

The temperature of the outflowing oil always increases with the speed, for the same adjustment of the regulator. It is also increased by diminishing the regulator opening, while the speed remains the same.

This is explained by considering that the power, and hence the energy transformed into heat, increases as well with the increase in speed as with the increase in torque, while, on the other hand, the increase in speed and pressure render it more difficult to maintain the water tightness of the gear wheels. There is, then, a diminution in the specific quantity per unit of transformed energy, or, to be exact, the quantity of the liquid, in which the generated heat is distributed, is diminished and, consequently, the temperature of the outflowing oil is increased.

Ratio between the Effective Power and the Mean Conventional Power.

For the purpose of explaining the operation of the brake, we calculated the mean conventional powers and their ratio to the measured ones (Table I). Evidently the value of the m.c.p. always

increases with the speed, since (even in experiments II and III, in which the pressure diminishes) the increase in speed is always such that p x n, to which the m.c.p. is proportional, increases, which is to say that both the effective power and the m.c.p. are increasing functions of the speed. If we now examine the ratio: $\frac{\text{effective power}}{\text{mean con. power}}$ we find that their mean value increases with the size of the regulator opening. This fact may be explained by saying that, with the increase of said opening, and hence with the diminution of the pressure, the manner of functioning of the brake is modified in the sense that increasing importance, with reference to the resisting torque is acquired by the eddies and the friction in comparison with the static reaction of the compressed oil on the gear teeth, so that the device acquires an increasing tendency to comport itself like a hydraulic brake.

The amplitude of the field, within which the values of said ratios (1.58 to 16.3) oscillate, is large, a range which gives some idea of the nature of the phenomenon just observed.

The fact that the ratios of experiment I decrease at first and then increase has no particular significance and is due only to the fact that the two functions m.e.p. and m.c.p. increase at different rates, although behaving in a very similar manner.

Conclusions.

The results obtained lead us to the conclusion that this brake has characteristics distinguishing it substantially from all other dynamometer brakes in use, while, on the other hand, it resembles

some of them within certain limits of functioning.

In this device a high degree of simplicity has been attained, not only in its functioning but also in the matter of regulating, which consists in opening a cock more or less, until the motive torque is balanced. This is quite an important characteristic in that it enables an operator to use the brake without special instructions. This simplicity of regulation, moreover, enables the brake to be made automatic by means of very simple devices, by keeping the torque or speed constant.

As regards its practical utilization, it is evident that its great accuracy enables the measurement of powers producing variable torques and speeds within extensive limits, so that, while at low speeds, it has the range of application common to Prony's brake and others derived from it, at high speeds it more closely resembles the electric and hydraulic brakes (dynamic brakes, Froude brake, Alden brake).

In fact, it may be said that the determining factors in the absorption of motive power are:

1) The sudden transformation, by shock, of kinetic energy, in flowing oil, by gear wheels (Segrè brake);

2) The friction of oil under pressure passing over the faces of gear wheels (Alden brake). It should, however, be here noted that, in the Alden brake, the oil pressure is automatically generated by the functioning of the brake;

3) The friction of the oil in the pipes and in the regulator;

4) The eddies and shocks of the oil set in rotation by the gear wheels (Froude brake);

5) The mechanical friction (bearings, gear teeth, stuffing boxes, etc.) (Prony brake and others derived from it);

6) Factors whose importance is largely determined by the speed, as well as by the pressure of the oil.

Regarding speed limits, it should be noted that the 1500 r.p.m. given in Table I, was only exceeded for the resistance of the armature of an electric engine in experimenting with which, a speed exceeding 2000 r.p.m. was attained for brief periods on several occasions.

Evidently, said limits are determined only by the resistance of the gear wheels (running in oil under pressure), with which, if made of suitable steel, we now readily obtain a linear velocity of 30 meters per minute at the pitch periphery.

The maximum torque depends, aside from the water tightness of the gear wheels and the brake case, also upon the kind of oil used and its temperature. The curves in Fig. 7 give an idea of the values attained and attainable with the model employed.

Lastly, regarding the powers capable of measurement by brakes of this type, we will only state that the theoretical powers increase, other things being equal, in proportion to the cube of the linear dimensions, while they are, on the other hand, proportional to the number of the torques of the gear wheels, as commonly arranged: that is, geared to the same central driving wheel or with a larger number of central coaxial wheels and constituting a single

brake, or in both ways at the same time.

In order to pass from the power P_1, given on Table I, to the values of P for a brake with dimensions λ times as large and with a number of gear wheels ν times as large, we only need to write (on the hypothesis that the ratios between the mean effective power and mean conventional power remain constant)

$$P = P_1 \lambda^3 \nu$$

Evidently these considerations cannot have an absolute value, but serve rather to indicate the nature of the attainable values, while, on the other hand, only a large and systematic series of experiments could lead to the predetermination of the forms and dimensions necessary for obtaining accurate results.

(Translated by National Advisory Committee for Aeronautics.)

TABLE I.

EXPERIMENT	I					II					
Observation	1	2	3	4	5	1	2	3	4	5	6
R.p.m.	220	400	600	1020	1400	400	600	800	985	1220	1450
Weight in grams	2692	3497	4300	7100	10500	2500	3000	3187	3387	3497	3550
Horse power	0.24	0.56	1.03	2.9	5.9	0.40	0.72	1.02	1.33	1.70	2.06
Regulator opening	2	2	2	2	2	5	5	5	5	5	5
Pressure (kg/cm^2)	5.4	8.5	11.5	16.9	21.7	5.6	5.6	5.3	5	4.8	4.8
Temperature of inflowing oil	27	26	26	27.5	27	27	26.5	27	27	26.5	26
Temperature of outflowing oil	30	32	37	42	42	31	31	31	32	35	35
Theoretic conventional power	0.11	0.31	0.63	1.58	2.78	0.20	0.30	0.39	0.45	0.53	0.64
Ratio between the measured power and the theoretic conventional power.	2.18	1.80	1.84	1.83	2.12	2.00	2.40	2.62	2.96	3.20	3.22
Remarks	..	..	..	..	..	..	..	..	..	..	..

TABLE I (Contd.)

EXPERIMENT	III				IV			
Observation	1	2	3	4	1	2	3	4
R.p.m.	400	800	1000	1500	400	600	800	1000
Weight in grams	1400	2386	2650	3000	1300	1705	2000	2200
Horse power	0.22	0.76	1.06	1.80	0.21	0.41	0.64	0.88
Regulator opening	10	10	10	10	15	15	15	15
Pressure(kg/cm ^{2})	1	0.9	0.9	0.8	--	--	--	---
Temperature of in-flowing oil	26	26.5	26	26.5	27	27	26.5	26.5
Temperature of out-flowing oil	28	29	32	37	28	28	28	30
Theoretic conventional power	0.04	0.06	0.08	0.11	--	--	--	--
Ratio between the measured power and the theoretic conventional power.	5.50	11.35	13.60	16.30	--	--	--	--
Remarks	..	..	..	..	During the whole experiment the manometer index hardly varied from zero.			

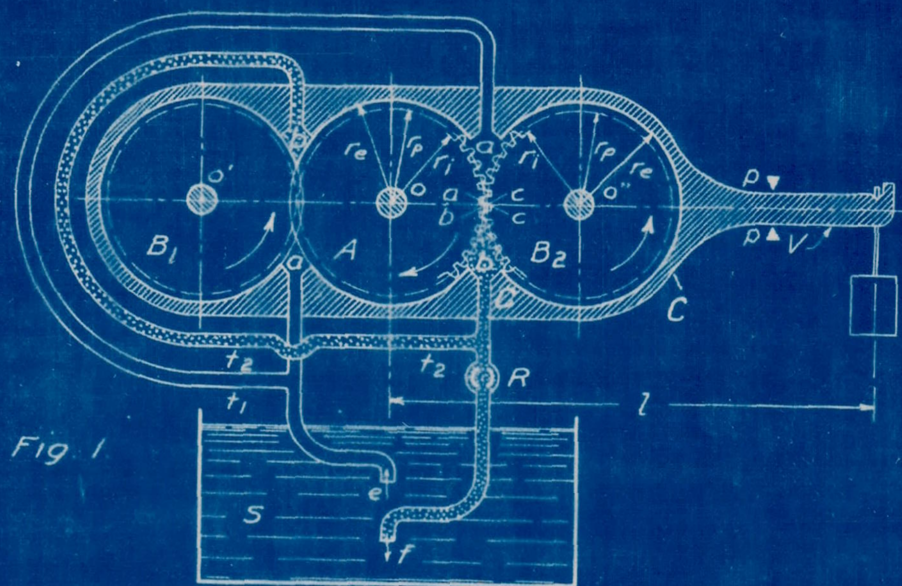

Fig. 1

Section YY

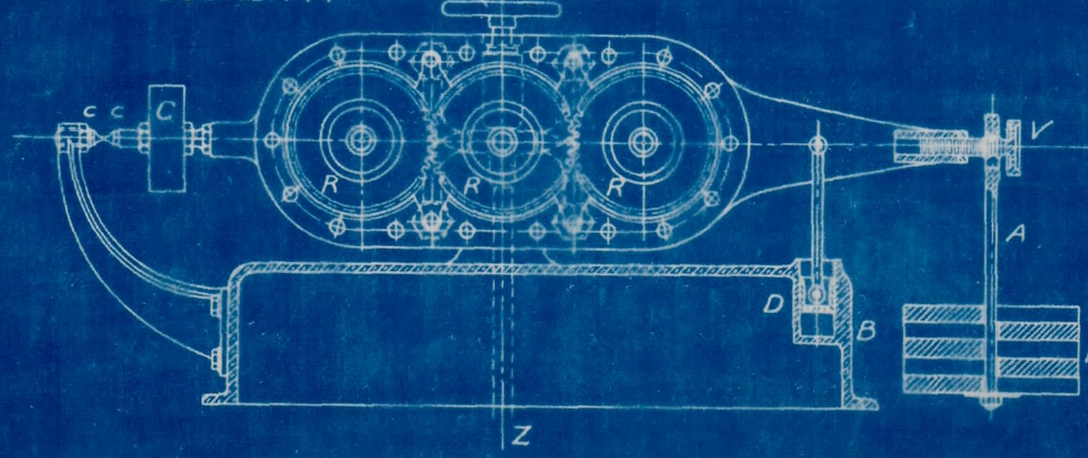

Fig. 2.

Section ZZ

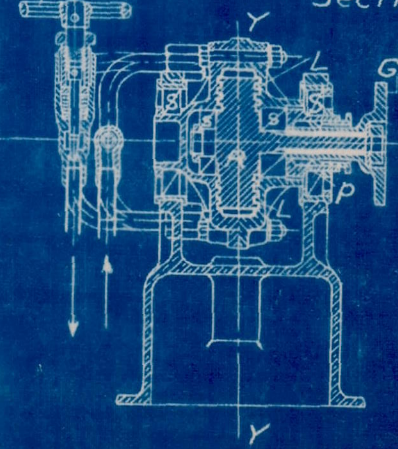

Fig. 3.

Figs. 1-2-3

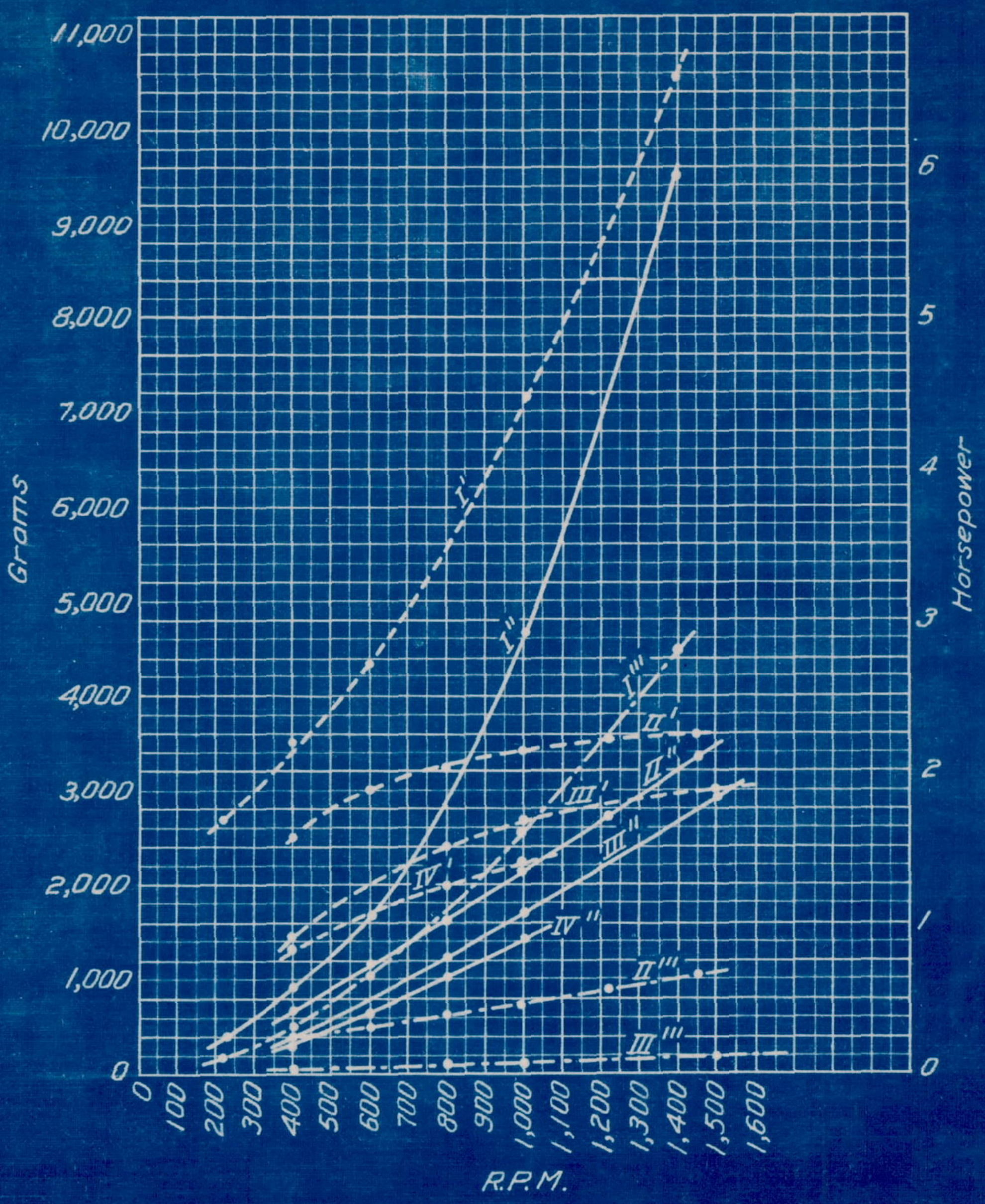

Fig. 4 - - - - - - - - Torque
——————— Effective horsepower
—·—·—·— Mean conventional horsepower

Fig. 4